EVAPORATION AND CONDENSATION OF R134a INSIDE A MICROFIN TUBE

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1. Summary

This paper reports on average heat transfer coefficients and pressure drops of oil-free R134a in flow-boiling and convective-condensation inside a microfin tube with a new cross-section profile. Data obtained for a smooth tube are also reported for comparison. Both tubes have the same outside diameter of 9.52 mm; they are horizontally operated and are heated/cooled by a water stream. The microfin tube is characterized by 82 sharp fins (apex angle of 40°) alternating with two different heights. Evaporation tests are carried out at a nominal temperature of 5 °C, for a mass velocity ranging from about 90 to 400 kg/(m^{2}s), inlet quality between 0.1 and 0.6, and quality change varying from 0.1 to 0.5; whereas for the condensation tests the nominal temperature is 35 °C, the inlet quality ranges from 0.8 to 0.4 and the quality change from 0.2 to 0.6.

2. Introduction

The use of microfin tubes in evaporators and condensers for refrigeration and air-conditioning applications has become a common technical solution because a relevant heat transfer enhancement is generally achieved with a moderate increase in pressure drops. Recently, the phase-out of CFC and HCFC refrigerants imposed by ozone layer depletion as well as by tightening of energy efficiency standards, has redirected research towards both testing alternative refrigerants and developing microfin tubes of advanced design. This technology was first reported in the open literature in the mid 1970's and since then an increasing number of papers on heat transfer characteristics of microfin tubes has been published and detailed literature reviews are available [1-4]. Several papers focused on the effects of various geometrical parameters such as tube diameter, spiral angle, fin height and shape, spacing between the fins and the number of fins [5-7] and proposed predictive correlations for both heat transfer and pressure drops [8-17]. However, due to the complexity of the physical phenomena involved in fluid-dynamics and heat transfer, experimental research, such as that here presented, is still the most reliable approach to the study of the performances of new microfin tubes and refrigerants.

In the following, results of an experimental analysis on flow boiling and convective condensation of R134a inside a o.d. 9.52 mm, horizontally operated microfin tube are presented and compared with the estimates obtained by recent correlations specifically proposed for this kind of tubes.

3. Experimental setup

A simplified schematic diagram of the experimental facility is shown in Figure 1. A more detailed description of the whole equipment was reported in a previous paper [18] therefore it will be summarized here. The rig consists of three circuits, namely, a sealed refrigerant circuit, a water circuit to evaporate or condense the refrigerant in the test section, and a chilled coolant (water-glycol solution) circuit. The main components of the refrigerant circuit are a boiler, the test section, a condenser, a gear pump and a filter dryer. The boiler is equipped with a heater consisting of three electrical cartridges of 1, 1.5 and 2.5 kW power. Liquid and vapor are drawn from it through two distinct lines; the liquid line is equipped with a subcooler, while a superheater is installed on the vapor line. On each line the refrigerant mass flow rate is measured by means of Coriolis-type meters. The liquid and vapor flow rates are controlled by precision metering valves. Downstream of the valves, vapor and liquid streams are mixed; the resulting two-phase mixture flows through a 1.5 m long calming section and then enters the test section. At the exit, refrigerant flows through a second calming section (1.8 m) and then is discharged to the condenser, which maintains the test section outlet pressure at a given value. Finally, the refrigerant is drawn from the condenser by a gear pump and is conveyed through a filter dryer to the boiler. The water loop sets the condition of the water entering the annulus side of the test section. It contains a centrifugal pump that circulates demineralized water, a magnetic flow meter that measures the flow rate, and a combination of a plate heat exchanger and an electrical heater controlling the water temperature. Finally, the chilled coolant circuit is filled with a water-glycol solution and it consists of a commercial refrigeration unit and a centrifugal pump. Such a circuit provides the cold medium circulating in the heat exchangers placed in the refrigerant condenser or mounted on both the refrigerant and the water circuits. The test section contains the tube to be tested. It consists of a double-pipe heat exchanger, divided into two identical subsections, where refrigerant flows inside the inner tube and water flows counter-currently in the outer annulus. The inner tube is made of copper with 9.52 mm o.d. and 1.3 m length. The distance between the inlet and discharge ducts of the jacket is 1.12 m which is assumed as the active heat transfer length for the subsection. At the exit of both subsections a sight glass made of 80 mm long, 8.5 mm i.d., pyrex smooth tube is mounted. These sight glasses are neither heated nor cooled. Two pressure-taps are located at the inlet and outlet of the first subsection and at the outlet of the second one, respectively. Both subsections are equipped with four T-type thermocouples to measure wall temperatures. The

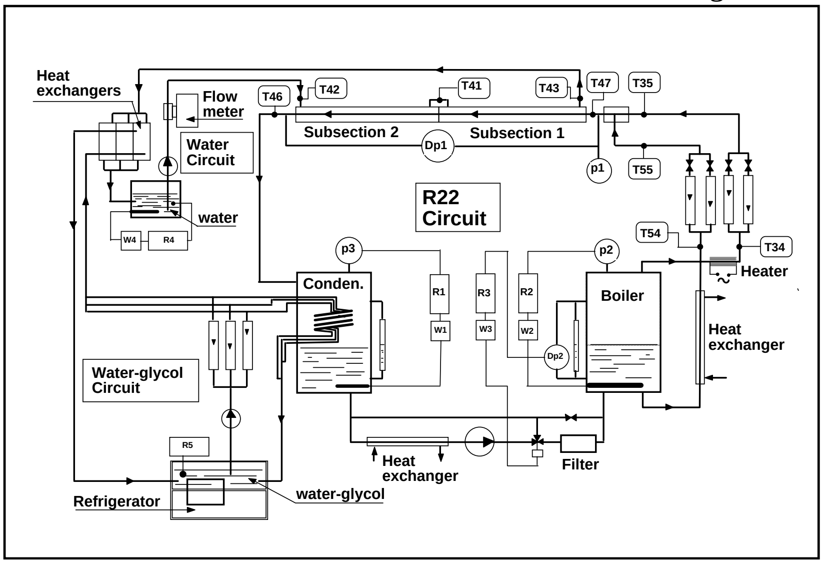


Figure 1. Schematic diagram of the experimental facility. Absolute and differential pressure transducers, temperature sensors, PID controllers and actuators are indicated respectively with p, Dp, T, R and W.

Parameter	HVA	smooth
Outside diameter [mm]	9.52	9.52
Max inside diameter [mm]	8.62	8.92
Bottom wall thickness [mm]	0.45	0.30
Higher fin height [mm]	0.20	-
Lower fin height [mm]	0.17	-
Apex angle	40°	-
Number of grooves	82	-
Helix angle	18°	-
Inside-surface area ratio	1.84	1

Table 1. Geometrical characteristics of tested tubes

thermocouples are placed in pairs on the top and at the bottom of the tube and are cemented in longitudinal grooves cut in the outside wall of the tube. Calming and test sections are thermally insulated by a 10 cm thick, glass-wool annulus. The microfin tube tested is Metofin 952-45HVA40/82 manufactured by Trefimetaux; for sake of simplicity, from now on we will refer to it as the HVA-tube. The main feature distinguishing it from other microfin tubes of new design is that fins alternate with two different heights, as reported in Table 1, together with the main geometrical parameters. For comparison data for the smooth tube of the same outer diameter are also listed.

4. Test procedures and data reduction

Signals from thermocouples and transducers are cyclically read by a data acquisition unit HP3497 and sent to an on-line PC. In order for all variables to be affected by similar RMS relative errors, the measurements of refrigerant temperature, pressure drop, refrigerant mass flow rate and water flow rate are based on 30, 50, 50 and 100 readings for cycle, respectively. Every experimental datum, instead, is obtained by averaging the measurements of ten cycles in order to reduce the influence of random errors and fluctuations. Finally, for every operating condition, more than ten experimental data are collected. The heat transfer coefficient is computed as follows. We assume that the refrigerant temperature varies linearly between the value T_{in}, measured at the entrance of the test section, and the value T_{out} computed at the exit as $T_s(p_s(T_{in})-\Delta p)$, where T_s is the function correlating the saturation temperature to the pressure, p_s the inverse function of T_s, and Δp the pressure drop measured along the test section. Then, for each subsection we calculate the mean refrigerant temperature $T_{r.m,i}$, the mean wall temperature $T_{w.m,i}$, the refrigerant to wall temperature mean difference $\Delta T_{m,i} = (T_{w.m,i}-T_{r.m,i})$, and the heat transfer coefficient $h_i=q_i/\Delta T_{m,i}$ where q_i is the mean heat flux based on a nominal inside area corresponding to the maximum internal diameter, i.e. the diameter at the root of microfins. Eventually, we compute the average heat transfer coefficient for the test section as the arithmetic mean of the subsection coefficients h_i. Relevant variables for the present investigation are affected by the following representative experimental uncertainties measured or estimated by a propagation error analysis: ±0.35% for the refrigerant mass flow rate, ±1.3% for the inlet quality, ±0.2 K between the refrigerant temperature and the saturation one, ±0.03 K between the wall and refrigerant temperatures with the refrigerant trapped in the test section and the water flowing, ±1.0% for the refrigerant pressure drop, ±1.0% for the water volume flow rate, ±0.02 K for the water temperature difference between the subsection inlet and outlet, ±1.4% for the heat rate, and ±7% for the average heat transfer coefficient, by assuming a temperature difference between wall and refrigerant equal to 2 K.

5. Results and discussion

In saturated flow boiling or convective condensation, for fixed test section configuration, i.e., dimension and shape of the cross section, length, orientation with respect to gravity, both pressure drop and average heat transfer coefficient depend on four independent variables, namely, total mass flow rate, temperature (or pressure), inlet thermodynamic quality and heat rate. Since the quality change along the test section depends linearly on heat rate, a different but equivalent parameterization can be obtained by substituting the former with the latter quantity in the list of the independent variables. Experimental tests were carried out at fixed saturation temperature, varying in turn the mass flow rate, the inlet quality and the quality variation, in order to assess clearly the influence of each variable on both heat transfer and pressure drop.

Evaporation tests are carried out at a nominal temperature of 5°C (±0.2 K) at the test section inlet, corresponding to a pressure of 0.350 MPa, for a mass flux ranging from about 100 to 340 kg/(m^2s), inlet quality between 0.25 and 0.70, and quality change varying from 0.10 to 0.70, whereas for the condensation tests the nominal temperature is 35 °C (±0.2 K) corresponding to a pressure of 0.887 MPa, the mass flux varies between 100 and 440 kg/(m^2s), the inlet quality ranges from 0.75 to 0.10 and the quality change from 0.10 to 0.70.

In the following subsections experimental data on evaporation and condensation are presented and discussed, respectively.

5.1 Evaporation

The influence of mass flux G and quality variation Δx both on the heat transfer coefficient h_b and pressure the drop Δp_b in evaporation was treated with some detail in a previous investigation [19] and trends will be summarized here. In particular, the heat transfer coefficient results an increasing function of the mass flux and the rate of increase is higher for lower values of the mass flux. This behavior could be explained in considering that the increase of G at constant Δx determines a concomitant growth of the heat flux: at low mass fluxes the contribution of saturated boiling causes the heat transfer coefficient to rise rapidly with the heat flux. When increasing G, convective evaporation takes place and the growth of the heat transfer coefficient is essentially due to that of the mass flux. Trends are similar to those obtained for the smooth tube of the same inner diameter, but h_b is always greater. An analogous consideration holds for the pressure drop. To summarize these results, in Figure 2 three parameters useful to compare the performance of the microfin tube with respect to the smooth one are reported as functions of the mass flux. Referring to $h_{b,s}$ and $\Delta p_{b,s}$ as the heat transfer coefficient and the pressure drop for the smooth tube, these three parameters are the enhancement factor $E_b=h_b/h_{b,s}$, the penalization factor $P_b=\Delta p_b/\Delta p_{b,s}$ and the efficiency index $E_i=E_b/P_b$. It is seen that E_b is a decreasing function of the mass flux and it seems to tend asymptotically to about 1.4, a value

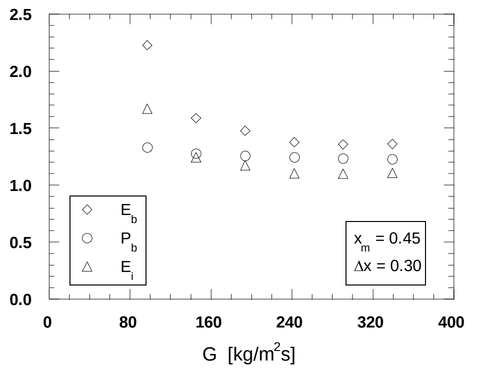
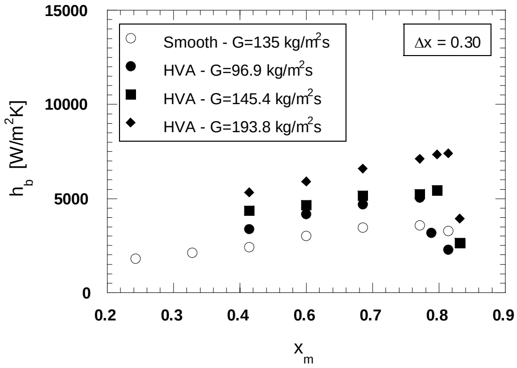


Figure 2. Enhancement factor, penalization factor and efficiency index, as functions of the mass flux in boiling.



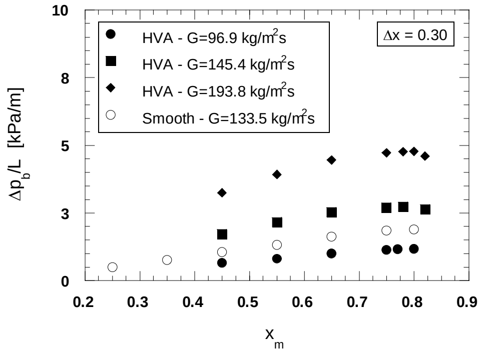


Figure 3. Flow-boiling heat transfer coefficient h_b versus average quality x_m for fixed mass flux G and quality change Δx.

Figure 4. Flow-boiling pressure drop $\Delta p_b/L$ versus average quality x_m for fixed mass flux G and quality change Δx.

smaller than the inside surface area ratio reported in Table 1. This means that the heat transfer enhancement is not simply justified by the area increase due to microfinning. Moreover, for this tube, the efficiency index E_i is just slightly greater than unity at the higher mass fluxes.

Finally, the influence of quality variation seems to be negligible in the considered range both for heat transfer and pressure drop.

The dependence of h_b on the average quality x_m for fixed mass flux and quality change is depicted in Figure 3. For the microfin tube, a distinct maximum in the heat transfer coefficient is observed at high average vapor quality, ranging from about 0.75 to 0.80. The corresponding value of x_m seems to increase slightly with the mass flux. The fall off in the heat transfer coefficient after the peak is caused by the dryout onset. In fact, heat transfer data relevant to the second subsection, not reported here, indicate that dryout occurs at a vapor quality of approximately 0.9. For the smooth tube, instead, a slightly marked maximum, located approximately at x_m=0.7, is observed. The comparison between the data for the HVA tube at G=145.4 kg/(m^{2}s) and those for the smooth one at about the same mass flux shows that microfinning seems to provide, in addition to a substantial heat transfer enhancement, a shift of the dryout occurrence in the region of higher qualities. This might be due to the effect of both capillarity and centrifugal force that causes the wall to be kept wet longer. Moreover, the heat transfer augmentation decreases with increasing vapor quality.

Regarding the pressure drop, Figure 4 plots Δp_b per unit length of the tube as a function of x_m at the same conditions considered for h_b. Generally speaking the pressure drop appears to increase with the average quality. However, at higher values of x_m, it may fall off toward the data point for the only vapor phase pressure drop.

Experimental data were compared with the predictions of some available correlations. Among the various models reported in the literature, the Cavallini et al. correlation [8] and the Kandlikar and Raykoff scheme of correlation [9] for the heat transfer coefficient and the Wang and Kuo correlation [10] for the pressure drop were considered. Starting with the heat transfer coefficient and referring to the whole data set, it was seen that the Cavallini et al. correlation strongly over predicts the data. Actually the mean deviation E is 74.5% and the standard deviation σ is 74.0%. The Kandlikar et al. scheme of correlation provides better estimates, with E=9.9% and σ=21.8%. In particular, Figures 5 and 6 depict, as examples, the comparison between predictions and data for h_b in two cases: in the former as a function of G at constant x_m and Δx; in the latter as a function

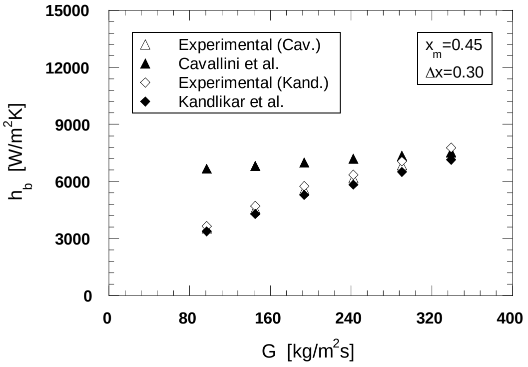


Figure 5. Flow-boiling heat transfer coefficient h_b versus mass flux G for fixed inlet quality x_{in} and quality change Δx. Comparison with correlations.

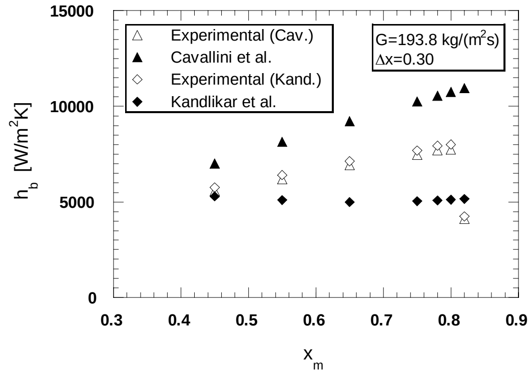


Figure 6. Flow-boiling pressure drop $\Delta p_b/L$ versus average quality x_m for fixed mass flux G and quality change Δx. Comparison with correlations.

of x_m, at fixed G and Δx. In considering these figures it is to be noted that experimental heat transfer coefficients are reported according to the definitions given for them in the correlations.

Inspection of Figure 5 reveals that the Cavallini et al. correlation over predicts the data particularly at the lower values of the mass flux, while the Kandlikar et al. scheme of correlation is able to reproduce the data trend with a good accuracy. The situation is reversed in considering Figure 6: in fact the Kandlikar et al. model predicts a heat transfer coefficient essentially constant with the average quality, while the Cavallini et al. estimate, though quantitatively incorrect, shows a behavior similar to data trend. In any case both correlations are unable to describe the dry out onset, clearly marked by the maximum in the trend of the experimental data. Regarding the pressure drop, the Wang and Kuo correlation provides a satisfactory agreement with the data set: the mean error is 24.4% and the standard deviation is 9.5%. In Figures 7 and 8 comparisons between prediction and data are reported at the same conditions considered in Figures 5 and 6 for the heat transfer coefficient. In this case the agreement is seen to worsen with increasing both mass flux and average quality.

5.2 Condensation

Convective condensation in the HVA-tube was extensively treated by the authors in [19]. In the following the most relevant observations will be recalled and a comparison with several correlations from literature will be presented. Figure 9 displays the average

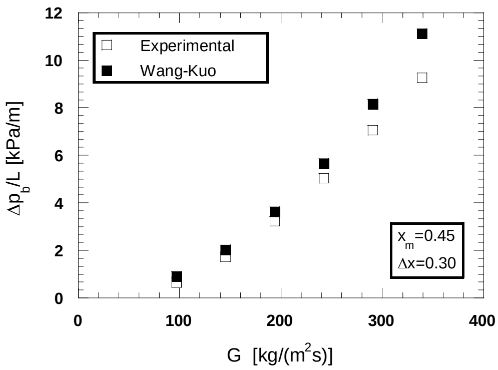


Figure 7. Flow-boiling pressure drop per unit length $\Delta p_b/L$ versus mass flux G for fixed inlet quality x_{in} and quality change Δx. Comparison with the Wang-Kuo correlation.

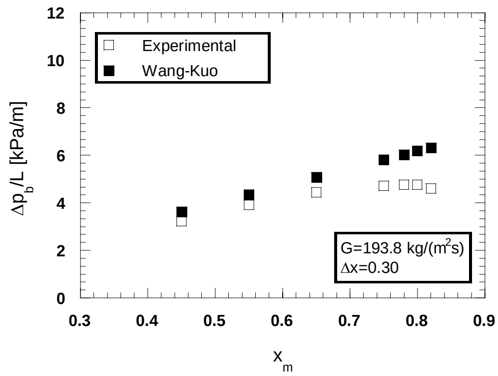


Figure 8. Flow-boiling pressure drop per unit length $\Delta p_b/L$ versus average quality x_m for fixed mass flux G and quality change Δx. Comparison with the Wang-Kuo correlation.

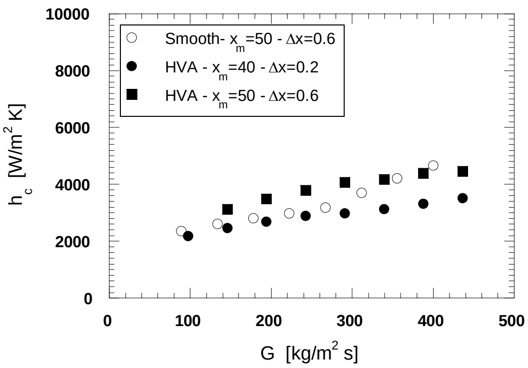


Figure 9. Condensation heat transfer coefficient h_c versus mass flux G for fixed inlet quality x_{in} and quality change Δx.

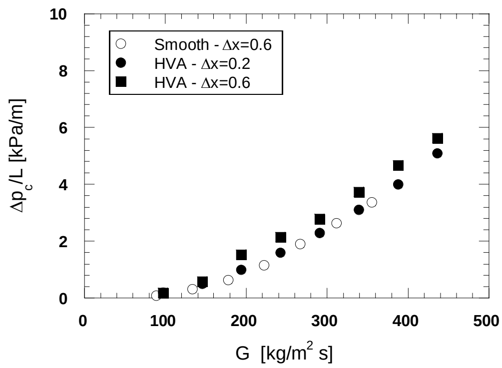


Figure 10. Condensation pressure drop $\Delta p_c/L$ versus mass flux G for fixed inlet quality x_{in} and quality change Δx.

heat transfer coefficient h_c plotted versus the mass flux G for the microfin and the smooth tube. As already observed, a mass flux variation at constant Δx is followed by a variation in the heat flux; for the data here reported, the average heat flux ranges from 1.5 to 13.2 kW/m^2. As expected, the heat transfer coefficient is an increasing function of G, but the trend for the microfin tube differs from that of the smooth tube. For the latter, data exhibit a linear-at-interval dependence on G with a change of slope approximately at G=250 kg/(m^{2}s). Supported by visual observations, we infer that in the first region, where heat transfer is weakly dependent on the mass flux, the flow is stratified, whereas it is annular when h_c starts to increase more steeply with G. On the contrary the trend for the HVA-tube flattens for high values of the mass flux and tends to the corresponding values of the smooth tube. Consequently the enhancement factor decays from about 1.4 to unity. In testing a similar tube, characterized by a lower number of fins (54 instead of 82), the authors found better performances at the same operating conditions, namely an enhancement factor varying between 1.25 and 1.5. This result is consistent with the findings of Yasuda et al. [5], who claim that a large number of fins decreases the heat transfer enhancement in condensation.

Regarding the influence of the average quality, it was observed that h_c increases with quality at constant mass flux and quality change. Instead the dependence on the quality change is less important. Analogous considerations hold for the pressure drop (Figure 10), which displays a relative increase comparable to that of the heat transfer coefficient: the penalization factor was found to vary between 1.4 and 2. Consequently the efficiency index results hardly greater than unity.

Experimental data were compared with results of some available models from literature. The correlation of Cavallini et al. [11], Kedzierski and Goncalves [12], Yu and Koyama [13] were selected for the heat transfer; the models of Cavallini et al. [11], Kedzierski and Goncalves [12], Haraguchi et al. [14], Nozu et al. [15] were considered for the pressure drop. Comparing experimental Nusselt numbers and predictions of the mentioned correlations, with reference to the whole data set, the mean error and the standard deviation have resulted respectively E=127.8% and σ=20.9% for the Cavallini et al. correlation, E=1.49% and σ=36.9% for the Kedzierski and Goncalves correlation, E=7.7% and σ=12.9% for the Yu and Koyama correlation. The ability in capturing trends may be verified from Figures 11 and 12. In the former the Nusselt number is reported versus mass flux; as already mentioned, experimental data was referred to the definition of the Nusselt number given in each model. It is evident that the results of the Cavallini et al. correlation strongly over predict data, even though the behavior seems to

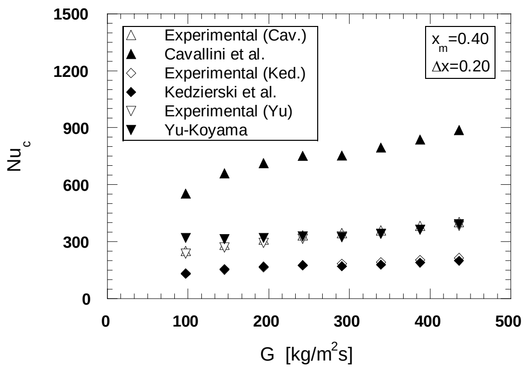


Figure 11. Condensation Nusselt number Nu_c versus mass flux G for fixed inlet quality x_{in} and quality change Δx. Comparison with correlations.

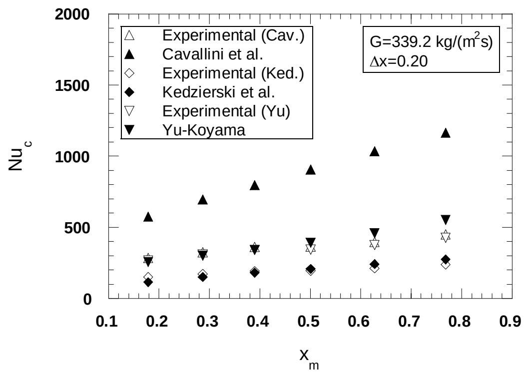


Figure 12. Condensation Nusselt number Nu_c $\Delta p_c/L$ versus average quality x_m for fixed mass flux G and quality change Δx. Comparison with correlations.

agree. The model of Kedzierski and Goncalves leads to a very good estimate, slightly under predicting at the higher values of G. On the contrary, the Yu and Koyama correlation tends to over predict data at the lower values of mass flux, but the agreement becomes excellent increasing G.

Figure 12 displays the Nusselt number as a function of the average quality: In this case all the correlations seems to reproduce the correct trend: the best agreement is provided by the model of Kedzierski and Goncalves, the worst is given by the Cavallini et al. correlation which strongly over predicts, while the estimate of Yu and Koyama worsen by increasing the average quality, over predicting the data. Attention will now turned to the pressure drop, starting from the mean deviation and the standard deviation of the selected models with respect to the whole data set: for the Cavallini et al. correlation $E=-11.2\%$ and $\sigma=23.0\%$; for the Kedzierski and Goncalves model $E=-11.9\%$ and $\sigma=14.1\%$; for the Haraguchi et al. correlation $E=5.9\%$ and $\sigma=16.2\%$; finally for the Nozu et al. model $E=64.0\%$ and $\sigma=22.1\%$. Comparison of trends is shown in Figures 13 and 14 where the pressure drop per unit length is reported as a function of the mass flux and the average quality, respectively. From the overall inspection of these figures, it is seen that the Haraguchi et al. correlation is the best predictor, while the models of Cavallini et al. and Kedzierski and Goncalves lead to equivalent estimates, generally under predicting the data. Moreover it is seen that the Nozu et al. correlation, even though its results are always much greater than measures, seems to capture the flattening trend at higher values of the average quality.

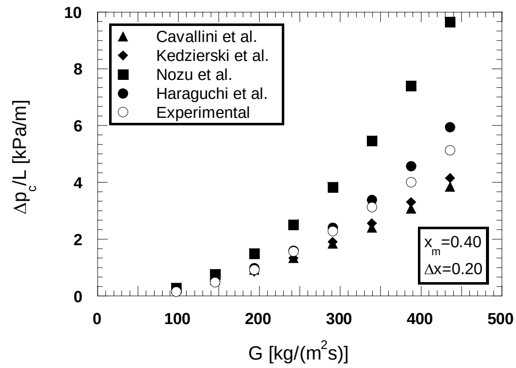


Figure 13. Condensation pressure drop per unit length $\Delta p_c/L$ versus mass flux G for fixed inlet quality x_{in} and quality change Δx. Comparison with correlations.

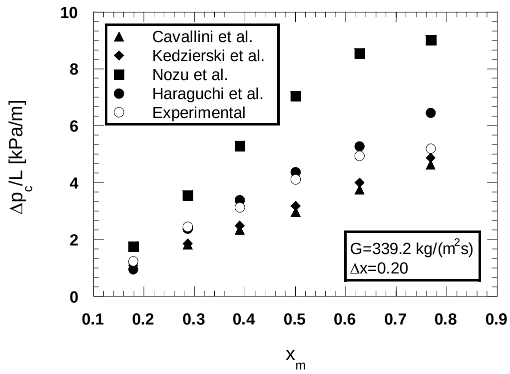


Figure 14. Condensation pressure drop per unit length $\Delta p_c/L$ versus average quality x_m for fixed mass flux G and quality change Δx. Comparison with correlations.

6. Conclusions

The present paper reports on results of an experimental investigation on flow-boiling and convective condensation inside a microfin tube of recent design. The data collected have been used to establish trends in heat transfer and pressure drop characteristics for flow boiling in this tubing. In order to highlight the effect of microfins both on heat transfer and pressure drop, data have been compared with those previously obtained on a reference smooth tube of the same outer diameter. From this investigation the following conclusions can be drawn.

In evaporation the microfin tube enhances heat transfer for the entire two phase region. Furthermore, this tubing displays a heat transfer enhancement factor always greater than the pressure drop penalization factor so that the efficiency index ranges in between 1.7 and 1.1; moreover the onset of dryout occurs at greater values of the average quality.

In condensation, instead, the large number of fins (82) seems to have a negative influence on heat transfer particularly for high values of mass flux. As a consequence, the efficiency index turns out to be hardly greater than unity.

Besides, data were compared with the predictions of some correlations selected from the literature. Regarding the heat transfer coefficient, the model of Kandlikar et al. showed the best agreement with the data in boiling, whereas the correlation of Kedzierski and Goncalves appeared to match the data very well in condensation. For the pressure drop, the model of Wang and Kuo, considered for boiling, provided a satisfactory agreement, while the Haragucy et al. correlation seemed to be the best predictor in condensation.

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